Experimental investigation of a micro jets - based cooling package for electronic applications

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Abstract

At present, most of the electronic components are cooled by means of heat sinks attached to them and by blowing air with fans. Unfortunately, this technique does not allow removing very high power without the heat sinks size becoming bulky or the fan becoming too large. An even bigger limitation of direct air-cooling appears when dealing with high heat fluxes, which are common since the chips’ size is becoming smaller by the day. Because reliability and speed of any chip depend on the working temperature, which normally must be below 120 °C, new techniques are needed to improve the heat removed per unit surface area and volume. Electronic components such as power IGBTs can require a heat removal rate up to 250 W/cm². An approach to accomplish such a formidable heat flux could be to use liquid micro jets sprayed on the cooling surface. The use of micro jets allows a fine control of the thickness of the liquid film that is deposited on the cooling surface. Studies have been conducted on this subject and heat fluxes as high as 300 W/cm² at relatively low flow rates, low surface temperatures, and associated pressure drops have been achieved.

The objective of the present work is to develop a closed loop cooling module, consisting of an orifice plate for creating micro-jets, a pump and an air cooled condenser. The impinging micro-jets allow high heat fluxes through the cooling surface at relatively low surface temperature. A small aluminum module, with pin fins on the outside, of only 1 dm³ in volume has been manufactured and tested. A diode was used as the heat source and it was cooled with micro-jets 140 µm in diameter impinging normal to the surface. The test fluid was water. Heat fluxes up to 300 W/cm² at a surface temperature, of 80 °C, were easily achieved. The effect of all major parameters, such as liquid flow rate, amount of non-condensibles present in the chamber, and airflow rate was investigated.

Introduction.

Spray and jet cooling have been studied for many years and they have been found to be capable of providing very high heat transfer coefficients. This nice feature has motivated their application to electronic cooling where, better ways of heat removal are needed to improve the reliability, which, in turn is inversely proportional to the chip’s temperature.

Spray cooling consists in spraying the hot surface with small diameter liquid droplets. The heat transfer mechanism is quite complicated and not completely understood. If the droplet frequency is high enough, the surface may be covered with a continuous film of liquid, in which case, the heat is mainly removed by convection and boiling from the surface. Several researchers have investigated the parameters affecting the performances of spray cooling, from droplet impinging velocity to droplet size (Choi and Yao, 1987), from surface temperature and wetting characteristics to liquid type and temperature (Labeish, 1994). It was found that heat fluxes as high as 15 MW/m² (Peterson, 1970) could be removed from the sprayed surface. At relatively low surface temperature of 130 °C heat fluxes on the order of 2.2 MW/m² were removed by Bonacina and Comini (1979).

When liquid jets are employed instead of sprays, a continuous layer of liquid covers the impinged surface and the heat transfer from the surface, at low surface temperatures, is principally convective and evaporative at the liquid – ambient interface. Boiling occurs when the wall superheat is well above the saturation temperature corresponding to ambient pressure. Even for this case a number of studies have been performed and the effect of various parameters such as jet diameter, velocity, liquid and surface temperature etc. has been investigated (Stevens and Webb, 1991). To obtain the highest heat transfer coefficients the liquid film deposited on the
surface has to be the thinnest. With this objective in mind a series of experiments has been conducted at UCLA
by Jiang and Dhir (2000) to find the best combination of jets’ diameter, spacing, and velocity.
The choice of employing jets instead of spray cooling finds its justification in the fact that the ratio between the
amounts of heat removed from the surface and the pumping power necessary to move the liquid is less for jets
than for sprays, as shown in table 1.
The objective of this work is to show that jet cooling can be employed for cooling of IGBT’s used in power
electronics. The jet-cooling concept is implemented in a closed functional system mimicking the conditions that
may exist in practice.

Cooling module and Experimental Apparatus description.

The cooling module, shown in Fig.1, consists of an aluminum box, 50 x 50 x 65 mm of internal dimensions and
3.175 mm of wall thickness. At the bottom, the box is closed with a 3.175 mm thick stainless steel plate. A 6.25
mm thick aluminum flange welded to the box walls provides the interface for the bottom plate and the space for
an o-ring.
Inside the container, a stainless steel orifice plate is installed on a support, which is located above the heat
source. The heat source consists of a diode used in current controlled mode to avoid high voltages. The diode is
mounted on a Direct Bond Copper (DBC) substrate layer, which is in turn glued on top of a G10 insulating base.
The diode is 8.68x4.97 mm in size. The electrical connections are provided by means of two copper rods 3.175
mm in diameter. Four threaded rods hold both the diode and the orifice plate assembly together and allow
adjustments of the relative distance between the diode and the orifice plate. The threaded rods are screwed into
the stainless steel plate, which forms the bottom of the box. The orifice plate, 0.5 mm thick, has 24 holes, which
are 140 µm in diameter, distributed on a rectangular array pattern with 2 mm spacing. A 3.175 mm OD stainless
steel tube connects the orifice plate to the outlet of the pump. The loop is closed with a 6.25 mm OD stainless
steel tube connecting the bottom plate to the pump inlet.
Aluminum pin fins 20 mm long and 3.175 mm in diameter are installed on the sides on a 45° staggered pattern
with both pitches equal to 10.16 mm. The fin tips are inserted in holes drilled into four aluminum plates, which
are successively welded at the corners and thus forming an external shroud. A small DC fan is mounted at the
bottom and pushes ambient air over the fins. The airflow is forced over the fins while filling the gaps in the
corners between the external box and the fin array. Figure 2 shows a sketch of the experimental apparatus.
K-type thermocouples are used to measure the air temperature at the inlet and outlet of the fin array, the inlet
water temperature, and the temperature of the environment in the chamber. Two RTD’s are used to measure the
temperatures on the top of the diode and on the back of the DBC.
An absolute pressure transducer is also installed to measure the pressure in the chamber. An outlet for the RTD’s
connecting wires and the electrical connections is provided on the top of the box. To check the liquid level inside
the box a short piece of tygon tubing is installed on the outside between the top and the bottom of the box.
Two pressure taps are present on one of the external plates and are connected to a differential pressure
transducer, which measures the pressure drop across the fins. A variable speed gear pump is used to pump the
liquid through the pipes and the orifice plate. The liquid flow rate is measured using a rotameter. The overall
dimensions of the whole module, including the fan, are: 100 x 100 x 130 mm.

Experimental procedure and data acquisition.

Before the experiment was started, the thermocouples were calibrated by submerging them in a pool of boiling
water and in an ice bath, whose temperatures were measured with a mercury thermometer with an accuracy of ±
0.1°C. The absolute pressure transducer (accuracy ±325Pa) reading was used to find the saturation temperature
of water and this was then compared with the one given by the thermometer. The two were found to be within ±
0.1°C.
The module was first charged with 40.8 ml of deionized water at room temperature. Subsequently the pump was
started and the flow rate was set to the chosen value. Thereafter, the internal pressure of the chamber was
reduced by means of a vacuum pump. At steady state chamber conditions, the vapor partial pressure was
calculated using steam tables from the temperature measured with the thermocouple present in the box, which was assumed to be equal to the vapor saturation temperature at the vapor partial pressure. Thereafter, power was supplied to the diode and data were recorded. The thermocouples, RTDs and box pressure readings were recorded with a 16-bit Strawberry Tree data acquisition system. The voltages supplied to the diode, to the fan and to the differential pressure transducer reading were recorded using a Fluke multimeter. The current through the diode was read directly from the power supply. Once all the values had become steady and had been recorded, either the power supplied to the diode or the fan speed was varied. Due to the unfortunate fact that thermocouples could not be used to measure the diode’s temperatures and that the RTD’s contact area was not negligible compared to the total area of the diode, a preliminary series of runs was performed, for every diode - DBC assembly tested, with one RTD attached on top of the diode and one on the back of the DBC layer. This was done to calibrate for the conductive resistance across the diode. The relationship between the temperatures on the front and back of the diode was linear as expected and it is shown in Fig. 3. While conducting the experiments reported in this work, the top RTD was removed to expose the whole top surface of the diode to the jets. The very small airflow, produced by a Flight II 80 DC fan manufactured by Comair-Rotron, 80x80x25 mm in size, could not be measured directly. To solve the problem, the relationship between the fan voltage and the pressure drop across the fins was first determined. Thereafter, air from a compressed air source, at a known flow rate measured with a Dwyer rotameter, was blown over the fins and the pressure drop was measured again. Finally, the relationship between airflow rate and fan voltage was established.

The minimum and maximum uncertainties were: for the power ±4.16% and ±8.13% of the measured value, for the heat flux ±4.16% and ±8.13%, for the thermal resistances ±4.16 % and ±8.72%, and for the heat transfer coefficient ±4.17% and ±8.58%.

Results and discussion.

As stated earlier, the purpose of this work was to investigate the performance of a jet-cooling module in a closed cycle. All the tests were conducted keeping the jet velocity approximately constant at 4.5 m/s. Figure 4 shows the heat flux at the diode surface as a function of the temperature difference between the top surface of the diode and the liquid. The effect of a higher mass fraction of air present in the chamber is clearly illustrated. For the same chip surface temperature of 80°C, a heat flux of 130 W/cm² as against 300 W/cm² can be removed when the mass fraction of air (m_air) is reduced from 95% to 13%. This also corresponds to a reduction of system pressure from 104 to 16 kPa. The mass fraction of air was obtained from:

\[ m_{\text{air}} = \frac{P_{\text{air}}}{P_{\text{air}} + (P_{\text{box}} - P_{\text{air}}) / P_{\text{box}}} \left( \frac{M_{\text{air}}}{M_{\text{water}}} \right) \]  (1)

where \( P_{\text{air}} \) and \( P_{\text{box}} \) are respectively the air partial pressure, and the total pressure in the chamber, and \( M_{\text{air}} \) and \( M_{\text{water}} \) are the molecular weights of air and water.

The upper limit for the data sets in Fig. 4 was either caused by the high current flow through the diode, as in the 13% \( m_{\text{air}} \) data set, or by the fan speed at the maximum value, as for the 97.5% \( m_{\text{air}} \), or by CHF conditions, as in the 28.7% \( m_{\text{air}} \) case. Furthermore, the data obtained by Jiang and Dhir using 150 µm jets 2 mm apart are plotted along in Fig. 4 to prove a satisfactory agreement. Figure 5 illustrates the effect of air content on the dependence of heat flux on temperature difference between the chip and the coolant, when the system pressure is maintained constant. To predict the module performance, the thermal resistances between the chip surface and the environment inside the module and between the latter and the air flowing on the outside was analyzed. The internal and external resistances were defined as

\[ R_{\text{int}} = \frac{T_{\text{chipfront}} - T_{\text{box}}}{Q} \]  (2a) and \[ R_{\text{ext}} = \frac{T_{\text{box}} - T_{\text{airin}}}{Q} \]  (2b)

where \( T_{\text{chipfront}} \) is the temperature of the wetted surface of the diode, \( T_{\text{box}} \) is the internal temperature of the module, \( T_{\text{airin}} \) is the ambient air temperature, and \( Q \) is the electrical power provided to the diode.
The external resistance was found to be unaffected by the presence of air inside the module, and depended only on the airflow provided by the fan indicating that the air side resistance dominated the condensation resistance on the inner wall of the box. The internal resistance increased when single phase force convection prevailed, with increasing mass fraction of air (Fig.7). The limit for zero mass fraction designates the resistance of the liquid film deposited on the chip surface by the jets, and excludes the diffusion resistance at the liquid – vapor interface.

It was also found that the difference between the box temperature and the saturation temperature, corresponding to the total pressure in the module, decreased with diminishing air fraction, and in the limit approached zero, a case in which the both temperatures are equal to the temperature at the evaporating interface (Fig.8).

With these data one can predict the module performance, and most importantly the chip temperature, which is the most crucial parameter affecting the chip’s life. The following example shows how to use the data to predict the performance of the system, given: an air flow rate of 32.5 m$^3$/hr, a power of 53 W to be dissipated, a partial pressure of air of 1250 Pa at room temperature, and an ambient temperature of 22 °C. The procedure is to determine $R_{ext}$ from Fig. 6 to find a value of 0.23 °C/W. Thereafter use eq. (2b) to find $T_{box}=34$ °C. Once $T_{box}$ is known the partial pressure of air must be corrected for the new box temperature to get 1300 Pa. Then Fig.8 should be used iteratively with the steam tables, $T_{box}$, eq. (1) and $P_{air}$ to obtain $T_{sat}(P_{box})-T_{box}=1.52$ °C, $P_{box}=5781$ Pa, and $m_{air}=0.317$. Lastly, using $m_{air}$ in Fig.7 $R_{int}$ can be found to be equal to 0.336 °C/W, and the chip temperature can be calculated using eq. (2a) to be 51.8 °C.

Another important aspect that must be considered is the ratio between the energy used for cooling the diode and operating the fan, and the heat removed from the chip. The pumping power was constantly around 3.8-5.1 W because the flow rate was kept constant, and the fan power varied from 0 to 2.2 W. From Fig. 9, it can be seen that the spray cooling becomes more effective as the heat removed from the diode increases: a ratio of 4.4% for the total power spent and the heat removed was the best result in this study. However this value is pessimistic as in several cases the power input to the chip was limited by restriction imposed by current rather than the critical heat flux. The efficiency of the gear pump used in this work was less than 1.5%. Improvement in pump efficiency can easily reduce the ratio of energy consumed to energy removed substantially below 1%.

Concluding Remarks.

The jet-cooling concept has been successfully implemented and applied to electronic cooling. A maximum heat flux of 300 W/cm$^2$ could be removed at a low surface temperature of only 80 °C when the mass fraction of air in the chamber was 13%. In the worst-case scenario of 95% air by mass, at the same surface temperature only 130 W/cm$^2$ could be dissipated. However, a heat flux of 204 W/cm$^2$ could be accommodated when the chip temperature reached 112 °C. The highest power density obtained was 0.129 W/cm$^3$, but it was limited by the current flow through the diode and not by the diode’s temperature. Even though this attempt proved to be quite successful, work must be done to reduce the external resistance and possibly also the internal one. This should allow smaller and less bulky dimensions of the module. Also some effort should be made towards a reduction in the power used for pumping the liquid in order to improve the process efficiency. However, considering the fact that very high heat fluxes are handled at low surface temperatures, the energy cost does not seem too high.

References.


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<th>Power Removed (W)</th>
<th>Power Removed (%)</th>
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Table 1 Comparison between different cooling methods by Jiang and Dhir.

Figure 1 Cooling module: Orifice plate and diode (left) and Top view (right)

Figure 2 Experimental apparatus schematic.

Figure 3 Relationship between front and rear temperatures of the diode.

Figure 4 Heat flux versus $T_{chipfront} - T_{liq}$. 
Figure 5 Effect of air on heat removal.

Figure 6 External resistance versus air flowrate.

Figure 7 Difference between the saturation temperature at box pressure and the actual box temperature.

Figure 8 Internal resistance as a function of the air mass fraction.

Figure 9 Process Effectiveness for a fixed liquid flow rate.